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(54) **Automotive heat exchangers**

(57) An automotive heat exchanger typically for use as a gas cooler in an automotive HVAC system having refrigerant operating in transcritical or supercritical state has an inlet header and an outlet header and at least two rows of tube lengths. A first row of tube lengths carry working fluid in a first direction away from the inlet header and a second row of tube lengths carry working fluid in a direction opposed to the first direction toward the outlet header. The headers are arranged in close proximity on the same side of the heat exchanger, an air gap separation or thermal insulator separation being provided between the headers.

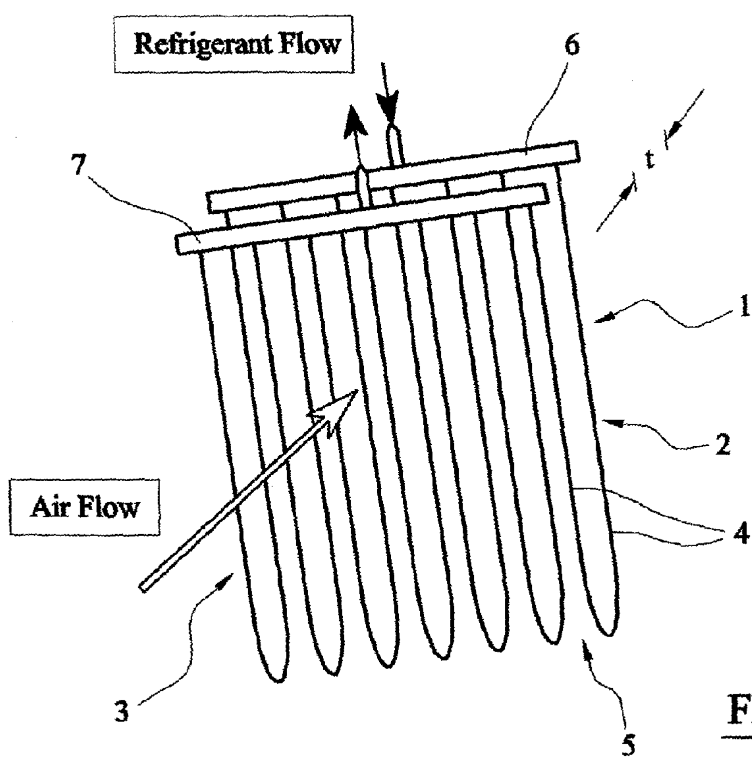

FIG. 1

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Description

[0001] The present invention relates to automotive heat exchangers and particularly, but not exclusively, to heat exchangers for automotive HVAC systems. In particular, the invention is suited but not limited to heat exchangers with refrigerants operating in a supercritical state and having a significant temperature slide through the heat exchanger.

[0002] Concerns of the adverse environmental impact of automotive HVAC system have seen efforts in developing alternative, more environmentally friendly automotive HVAC systems. The conventional chlorofluro-carbon (CFC) refrigerant has been known to be damaging to the ozone layer. Its replacement, hydroflurocarbon (HFC) refrigerants, is not ozone depleting, but has significant global warming impacts. This has lead to it being stipulated as an undesired gas whose usage should be monitored in the Kyoto protocol in 1995. Since then many countries have had legislation to phase out the usage of HFC over a number of years or to penalise its usage financially by levying heavy environment-protection taxes.

[0003] Many efforts have been made to develop more environmentally friendly automotive heat exchangers. An improved system has now been devised.

[0004] According to a first aspect, the present invention provides an automotive heat exchanger comprising:

an inlet header;
an outlet header;
at least two rows of tube lengths; a first row of tube lengths carrying working fluid in a first direction away from the inlet header; and a second row of tube lengths carrying working fluid in a direction opposed to the first direction toward the outlet header.

[0005] The heat exchanger is particularly suited to use in an automotive HVAC system based on the use of the natural gas CO_2 as the refrigerant. It has been proved in both bench tests and prototype vehicles that such CO_2 automotive HVAC system can obtain better cooling and heating capacities with also higher energy efficiency, over a wide range of operating conditions.

[0006] In a preferred embodiment the inlet header and outlet header are arranged in close proximity at the same side of the heat exchanger. The inlet and outlet headers are preferably arranged in substantial thermal insulation separation to inhibit heat transfer between the inlet and outlet headers. In order to effect this, an air-gap separation may be provided between the inlet and outlet headers. Additionally or alternatively a thermal insulator element (or elements) may be provided between the inlet and outlet headers.

[0007] It is preferred that a return portion of the heat exchanger is provided, arranged to permit return of working fluid from the first tube lengths to pass in the opposed direction along the second tube lengths. The return portion of the heat exchanger beneficially comprises return portions of individual tubes, arranged to permit return of working fluid from the first tube lengths to pass in the opposed direction along the second tube lengths. Alternatively, the return portion may comprise a return header arrangement, arranged to permit return of working fluid from the first tube lengths to pass in the opposed direction along the second tube lengths. In this embodiment, the return header arrangement may comprise a working fluid reception header and a dispatch header and interconnected fluid communication means.

[0008] It is preferred that the tube lengths in each row are spaced to define a flow way for a second heat exchange fluid, the second heat exchange fluid preferably being air.

[0009] Beneficially, the rows of tube lengths are preferably spaced in the direction of travel of the second heat exchange fluid through the heat exchanger.

[0010] According to a second aspect, the invention provides a heat exchanger tube for an automotive heat exchanger, the heat exchange tube being a substantially flat tube with internal fluid bores, and including a first tube length and a second tube length substantially parallel to and spaced from the first tube length and a return tube portion connecting the two spaced tube lengths, the tube width being substantially greater than the tube depth.

[0011] In a preferred embodiment, the invention provides a heat exchanger comprising a gas cooler for an automotive HVAC system, the gas cooler having a refrigerant fluid operating in transcritical or supercritical state, the gas cooler having an inlet header and an outlet header arranged in close proximity on the same side of the gas cooler an air gap separation or thermal insulator separator being provided between the headers; at least two rows of the tube lengths being provided, a first row of tube lengths carrying refrigerant in a direction away from the inlet header, and a second row of tube lengths carrying working fluid in a direction opposed to the first direction, toward the outlet header; a return portion being provided to permit return of the refrigerant from the first tube lengths to pass into the second tube lengths.

[0012] The invention will now be further described, by way of example only and with reference to the accompanying drawings, in which:

Figure 1 is a schematic representation of a first embodiment of automotive heat exchanger in accordance with the invention;

Figure 2 is a schematic representation of a second embodiment of heat exchanger in accordance with the invention;

Figure 3 is a schematic part section view of a heat exchange tube for use in accordance with the invention in a heat exchanger for example as shown in Figure 1;

Figure 4 is a alternative perspective view of the heat exchange tube of Figure 3;

Figure 5 is a view similar to the view of Figure 4 showing the heat exchange tube together with fin/ airway arrangement;

Figures 6 to 8 are graphic representations comparing the performance of designs in accordance with the invention with a single row, multiple pass parallel flow gas cooler (similar to a conventional parallel flow condenser) of similar packaging space and under the same refrigerant mass flow rate and airflow conditions.

[0013] Due to the particular thermophysical properties of CO_2, namely its low critical temperature of about 31 °C, a CO_2 HVAC system will most of the time operate in a transcritical cycle during air conditioning mode. The refrigerant CO_2 will therefore typically absorb heat from the air to be conditioned at a subcritical state, by means of an evaporator, and then emit the heat into the ambient air at a supercritical state, by means of a gas cooler.

[0014] Operating in supercritical conditions, the CO_2 refrigerant experiences a significant temperature slide, changing from as high as 160°C to as low as 35°C, flowing through the gas cooler. It is well established that CO_2 system performance, in terms of both the coefficient of performance (COP), (defined as the ratio of cooling capacity to the power consumed for delivering the cooling effects) and cooling capacity, is critically sensitive to the refrigerant approach temperature at the gas cooler exit and pressure loss through the gas cooler. The objective of a good gas cooler design is therefore to achieve the lowest approach temperature at possible minimum pressure loss, or the best combination of the two, to obtain the best system performance.

[0015] A constraint of gas cooler design is that its physical dimension is usually limited by the under bonnet packaging space. Manufacturer requirements mean that the CO_2 gas cooler must be accommodated within the same packaging space as the condenser in prior art systems. In this spirit, many current gas cooler designs follow the current best condenser design practice of one-row, multipass parallel flow. This however does not take into account the significant difference between a conventional condenser and the gas cooler. The conventional refrigerant in a condenser flows mainly in two-phase state, and thus the refrigerant temperature over the majority of condenser is almost constant (changes very slightly due to a small pressure drop in the condenser). In a gas cooler, the refrigerant flows in supercritical state without phase change and thus a significant temperature slide occurs over the refrigerant flow path. It follows that the optimal design for the gas cooler may be different from that of conventional condenser if this significant temperature slide is properly taken into account. The other disadvantage of gas cooler following a conventional one-row multipass condenser design is that it will result in very uneven air-off temperature from the gas cooler, which in extreme cases can lead to the heating up of part of the downstream vehicle radiator instead of cooling it, which could especially be a problem for unified gas cooler (condenser) and radiator.

[0016] The present invention provides the designs that take into account of the particular characteristics of heat exchangers operating in supercritical state.

[0017] Referring to Figure 1, the preferred design is a cross-counter flow, two tube length row, two pass heat exchanger 1. The two rows 2, 3 of the tube lengths 4 are formed from a continuous single tube, turned and twisted back at a return portion 5 of the heat exchanger for the two rows of tube lengths to be parallel and in line. The return portion 5, as can be seen from the drawings, lies in the same plane, and has the same height (depth) dimension as the tube lengths as the tube lengths in rows 2, 3. This provides enhanced flow characteristics. The first row of tube length is connected to the collection header 6 and the second row connected to the distribution header 7. The two headers 6,7 are located on the same end of the heat exchanger and they are separated with a small gap (t) to prevent heat conduction between them. Additionally or alternatively, a non-conductive connection may be provided. A common airway 9 spanning the both tube rows is brazed to the two tube rows (as shown most clearly in Figure 5). In order to minimize the longitudinal heat conduction through the airway 9, the bridge region 10 of the airway falling between the regions of the two tube rows is cut with a region 11 of multiple narrow and long louvers or slits, which also promote local heat transfer to the air stream. In order to enhance structural strength and the return position 5 of the heat exchanger, the return portion of the tubes can be brazed to a plate. Apart from offering better performance, as referred to below, the arrangement avoids using a return header (or headers) and related connectors, which aids in reducing cost and minimizing brazing problems (such as leakage and blockage, among the tube, connector and header joints). Figures 3 and 4 show a exemplary tube including forward and return lengths 12, 13 and tube return portion 15.

[0018] A second embodiment of the present invention (shown in Figure 2) is a cross-counter flow, multiple row heat exchanger with headers (6, 7, 26, 27) at both ends. This design avoids the need to twist and turn the tubes and also provides flexibility of multiple pass arrangements in each individual row. Again it is crucial to have headers (6, 7) that connect to each row of tubes separated from each other with a small gap (t). Common airways cut with narrow long louvers or slits between the bridge regions are brazed to the multiple rows of tubes as shown for the embodiment of Figures 1 and 5.

[0019] Simulations have been used to compare the performance of the current design with the design of a single row, multiple pass parallel flow gas cooler (corresponding to a conventional parallel flow condenser) of

the same packaging space (length by width by depth) under the same refrigerant mass flow rate and air flow conditions. Figures 6 to 8 show the results of the performance comparison. It can be seen heat transfer rate can be improved by 15% with only a small increase in pressure drop. The increase in small pressure drop is mainly due to the header loss which can be further optimized to minimize the pressure drop.

Claims

1. An automotive heat exchanger comprising:

an inlet header;
an outlet header;
at least two rows of tube lengths; a first row of tube lengths carrying working fluid in a first direction away from the inlet header; and a second row of tube lengths carrying working fluid in a direction opposed to the first direction toward the outlet header.

2. An automotive heat exchanger according to claim 1, wherein:

i) the inlet header and outlet header are arranged in close proximity at the same side of the heat exchanger; and/or

ii) the inlet and outlet headers are arranged in substantial thermal insulation separation to inhibit heat transfer between the inlet and outlet headers.

3. An automotive heat exchanger according to claim 2, wherein:

i) air-gap separation is provided between the inlet and outlet headers; and/or

ii) a thermal insulator element is provided between the inlet and outlet headers.

4. An automotive heat exchanger according to any preceding claim wherein a return portion of the heat exchanger is provided, arranged to permit return of working fluid from the first tube lengths to pass in the opposed direction along the second tube lengths.

5. An automotive heat exchanger according to claim 4, wherein the return portion of the heat exchanger comprises return portions of individual tubes, arranged to permit return of working fluid from the first tube lengths to pass in the opposed direction along the second tube lengths.

6. An automotive heat exchanger according to claim 5, wherein the return portion comprises a return header arrangement, arranged to permit return of working fluid from the first tube lengths to pass in the opposed direction along the second tube lengths, preferably wherein the return header arrangement comprises a working fluid reception header and a dispatch header and interconnected fluid communication means.

7. An automotive heat exchanger according to any preceding claim, wherein the tube lengths in each row are spaced to define a flow way for a second heat exchange fluid, the second heat exchange fluid preferably being air.

8. An automotive heat exchanger according to claim 7, wherein the rows oftube lengths are spaced in the direction of travel of the second heat exchange fluid through the heat exchanger.

9. An automotive heat exchanger according to any preceding claim, wherein heat transfer fins extend in the space between adjacent tubes in a respective row, preferably wherein heat transfer fins extend between tubes in adjacent rows, preferably wherein the heat exchanger fins have a thermal flow restriction inhibiting heat transfer via the fins across the space between the adjacent rows of tubes, preferably wherein the thermal flow restriction comprises one or more slits or louvres provided through the fin in the portion between the rows of tubes.

10. An automotive heat exchanger according to any preceding claim, wherein the working fluid carried in the heat exchanger is a fluid operating in supercritical state, preferably wherein the working fluid carried in the heat exchanger is CO_2 refrigerant.

11. An automotive air conditioning condenser (gas cooler) comprising a heat exchanger according to any preceding claim.

12. An automotive HVAC system including a heat exchanger according to any preceding claim.

13. A heat exchanger tube for an automotive heat exchanger, the heat exchange tube being a substantially flat tube with internal fluid bores, and including a first tube length and a second tube length substantially parallel to and spaced from the first tube length and a return tube portion connecting the two spaced tube lengths, the tube width being substantially greater than the tube depth.

14. A heat exchanger tube according to claim 13, wherein the first and second tube lengths are spaced in the direction of tube width.

15. A heat exchanger tube according to claim 13 or claim 14, wherein the return portion of the tube lies in substantially the same plane, and has substantially the same height (depth) as the first and second tube lengths.

16. A heat exchanger comprising a gas cooler for an automotive HVAC system, the gas cooler having a refrigerant fluid operating in transcritical or supercritical state, the gas cooler having an inlet header and an outlet header arranged in close proximity on the same side of the gas cooler an air gap separation or thermal insulator separator being provided between the headers; at least two rows of the tube lengths being provided, a first row of tube lengths carrying refrigerant in a direction away from the inlet header, and a second row of tube lengths carrying working fluid in a direction opposed to the first direction, toward the outlet header; a return portion being provided to permit return of the refrigerant from the first tube lengths to pass into the second tube lengths.

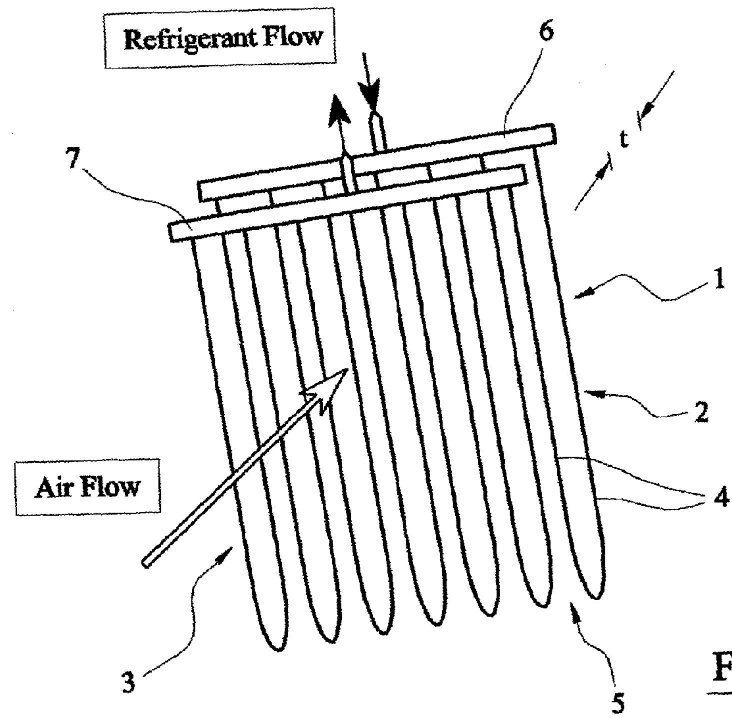

FIG. 1

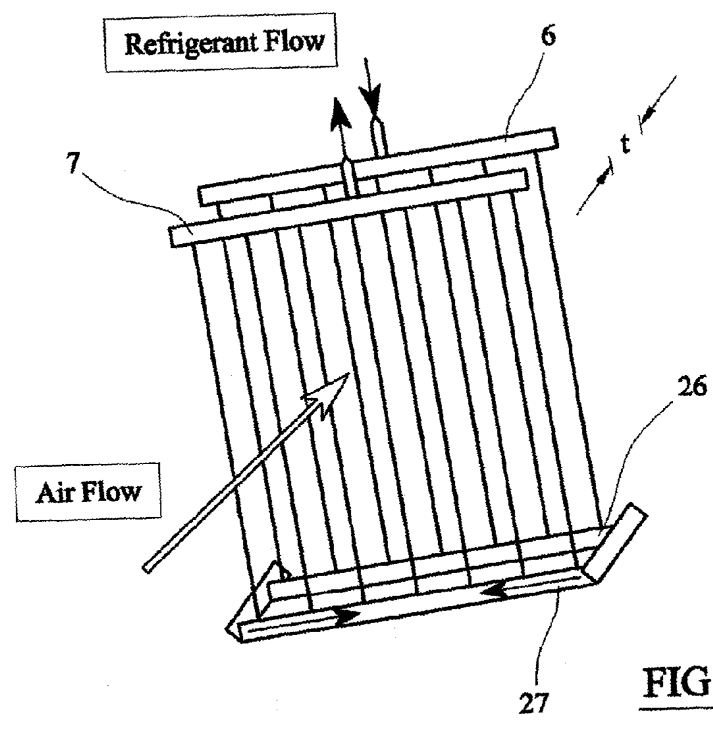

FIG. 2

Newly filed

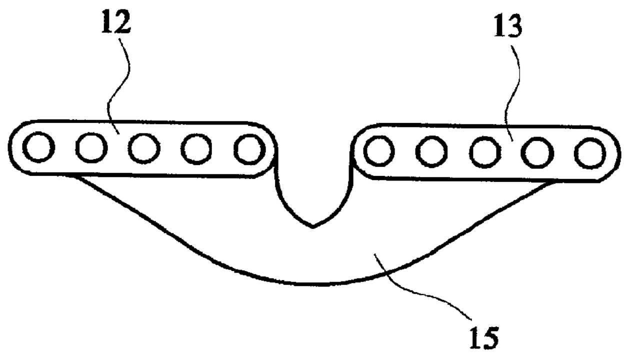

FIG. 3

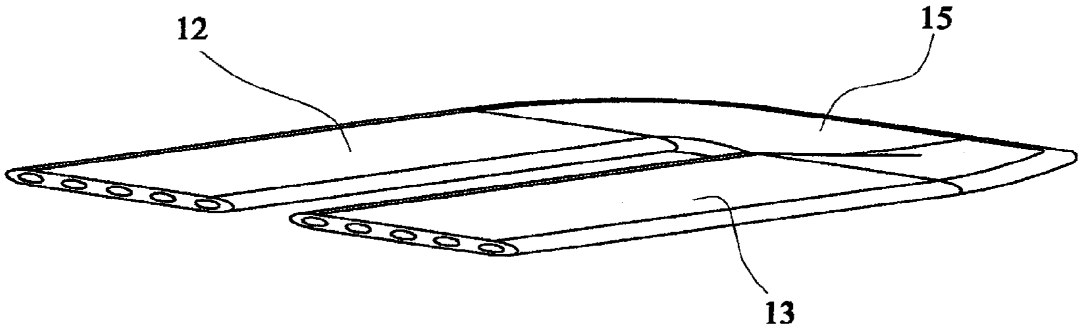

FIG. 4

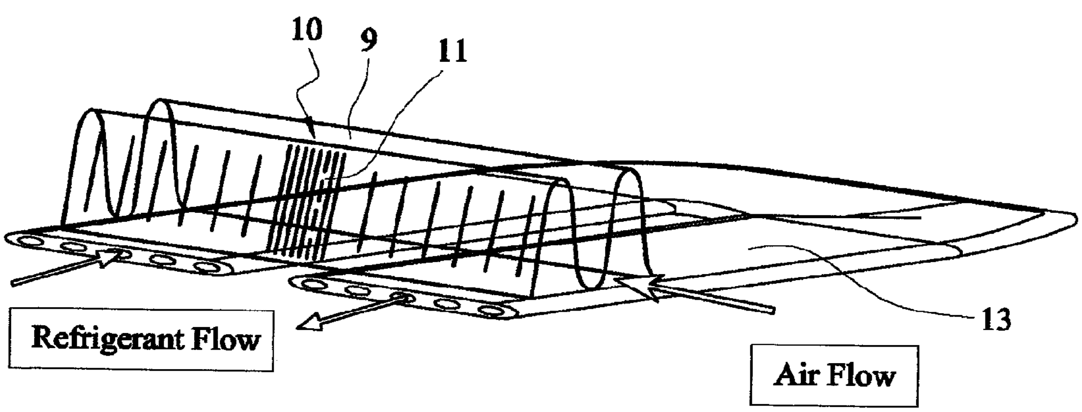

FIG. 5

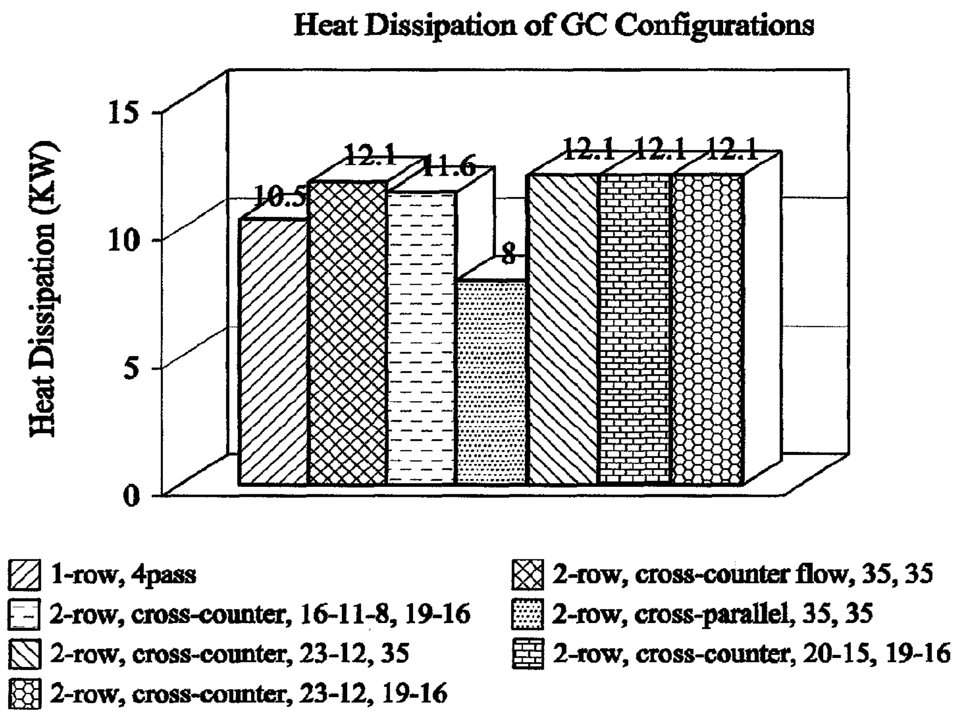
Heat Dissipation of GC Configurations
Heat Dissipation (KW)
15
10
5
0
10.5
12.1
11.6
8
12.1
12.1
12.1
1-row, 4pass
2-row, cross-counter flow, 35, 35
2-row, cross-counter, 16-11-8, 19-16
2-row, cross-parallel, 35, 35
2-row, cross-counter, 23-12, 35
2-row, cross-counter, 20-15, 19-16
2-row, cross-counter, 23-12, 19-16

FIG. 6

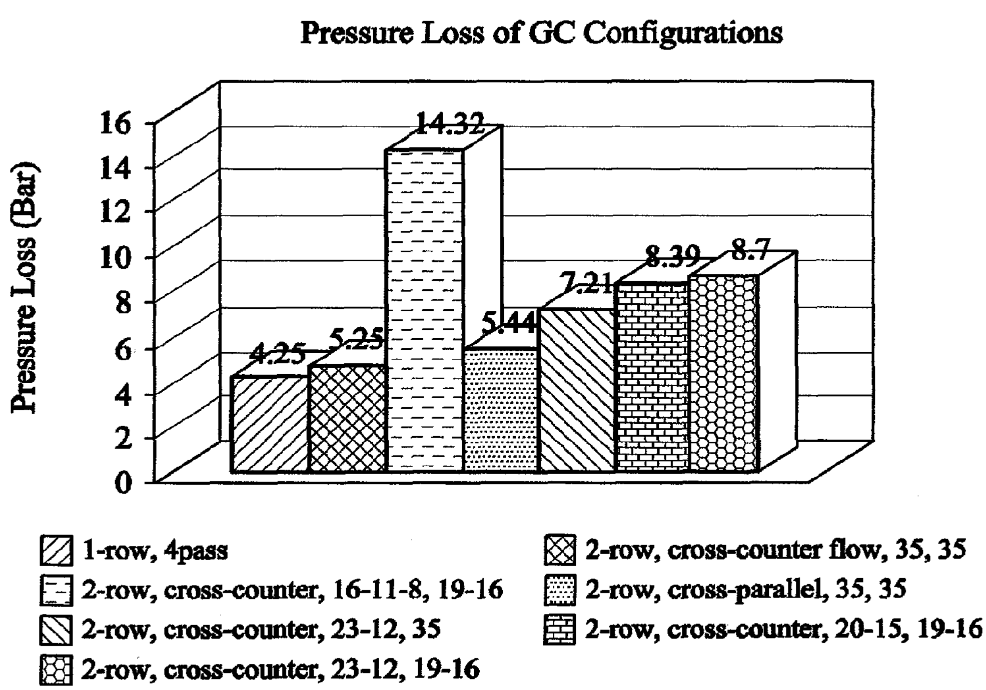
Pressure Loss of GC Configurations
Pressure Loss (Bar)
16
14
12
10
8
6
4
2
0
4.25
5.25
14.32
5.44
7.21
8.39
8.7
1-row, 4pass
2-row, cross-counter flow, 35, 35
2-row, cross-counter, 16-11-8, 19-16
2-row, cross-parallel, 35, 35
2-row, cross-counter, 23-12, 35
2-row, cross-counter, 20-15, 19-16
2-row, cross-counter, 23-12, 19-16

FIG. 7

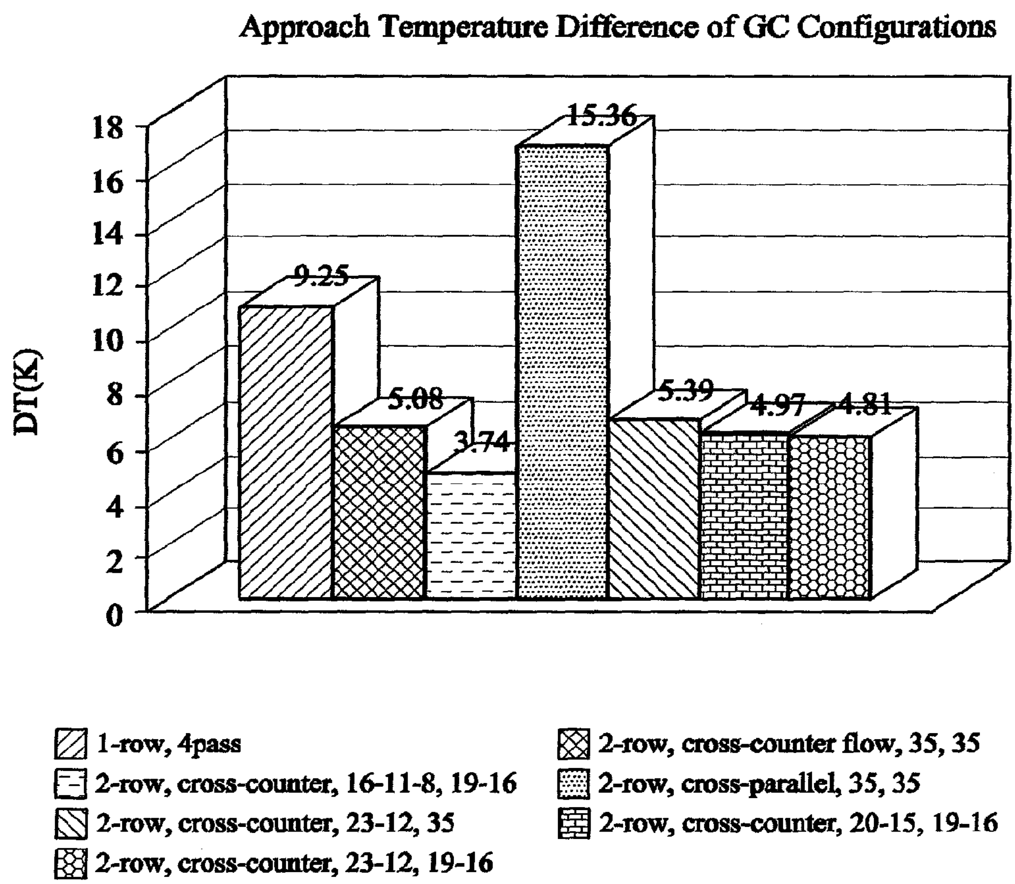
Approach Temperature Difference of GC Configurations
DT(K)
18
16
14
12
10
8
6
4
2
0
9.25
5.08
3.74
15.36
5.39
4.97
4.81
1-row, 4pass
2-row, cross-counter flow, 35, 35
2-row, cross-counter, 16-11-8, 19-16
2-row, cross-parallel, 35, 35
2-row, cross-counter, 23-12, 35
2-row, cross-counter, 20-15, 19-16
2-row, cross-counter, 23-12, 19-16

FIG. 8